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THE SERVICE LIFE OF BALL BEARINGS

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ABSTRACT: The durability and service life of ball bearings cannot be established dependably either through tests on conventional samples of materials or through mathematical calculations. Only tests performed on complete bearings will yield results. Equations were established on the basis of such tests that can be used to determine bearing life under purely radial loads, composite loads, and under alternating load intensities over time. The service life of a ball bearing exposed to axial or composite loads is determined in part by the greater ball pressure that results and in part by the fatigue phenomena which are subject to change, compared to conditions prevailing under purely radial loads.

The determination of the durability of ball and roller bearings can be considered one of the most important tasks of modern anti-friction bearing technology. A ball bearing has much more complicated structure than most other machine components, and it does not even require very deep penetration of the existing problems to arrive at the conclusion that the calculation of the required and cost-effective dimensions of ball bearings involves very great difficulties indeed.

The load capacity of a ball bearing cannot be compared to the durability of an ordinary machine component which, as a rule, is simply dimensioned so that the load stresses on the material remain within a certain margin which is determined by the precisely known fracture strain, elastic limit, or fatigue limit of the material in question. While it is possible that a ball bearing is subject to a certain fatigue limit which has a certain relationship to the so-called permissible load, and while such a fatigue limit can be considered applicable to ball bearings, experience has shown that this fatigue limit cannot be used as input data for a determination of ball bearing dimensions. It is a strange

* Numbers in the margin indicate pagination in the foreign text.

but true fact that it is often necessary to exceed the fatigue limit in order to obtain economical ball bearing dimensions.

The first approach to the solution of this problem, as in many problems encountered in engineering, is to perform a theoretical calculation which is frequently based on experimental data. In the early days of the ball bearing industry the derivation for the calculation of deformations and stresses upon contact between the curved surfaces of elastic bodies, developed by the German physicist Hertz [1], was already available. This derivation is based on a number of simplifying stipulations which will not yield very accurate approximation values, for instance when calculating the deformations. Moreover, recent investigations made at the A. -B. Svenska Kullager-Fabriken (SKF) [2, 3] have proved through calculations and experiments that the Hertzian formulae will not yield a generally applicable procedure for calculating the material stresses, nor the load capacity and service life of ball and roller bearings. Apparently this inherent weakness of the Hertzian formulae has not been given sufficient attention by the ball bearing industry since there is still a tendency to use these formulae beyond their area of applicability.

The necessity of practical experiments was recognized already by Stribeck [4] who laid the groundwork of practical ball bearing technology through his well-known and excellent investigations. However, it was not until recent years that a better understanding has developed into the fact that a relationship exists between load and service life. H. Gartner [5] has published some results of his investigations into this relationship, but he made no attempt to establish a definite law for this relationship. It is certain that the results available to him were too small in number and too irregular to derive such a law. Subsequently, F. Symanzik [6] proposed a procedure for calculating the load-bearing capacity of a ball bearing which was based primarily on a graphic technique. Unfortunately, he failed to state the supporting information for his rules, and the law in question was not substantiated through calculations so that the use of the calculation procedures published by him does not permit, for instance, to obtain an evaluation of the change in the service life of a certain ball bearing that is caused by a change in load.

As a result of the paramount importance of this problem to ball bearing technology, comprehensive in-house studies were performed at SKF in order to

find the law that describes the change in service life that is caused by changing load, r.p.m., bearing dimensions, and the like. There was only one possible approach: tests performed on complete ball bearings. It is not acceptable to perform theoretical calculations only, since the actual stresses that are encountered in a ball bearing cannot be determined by mathematical means. Moreover, conventional fatigue tests of the materials cannot be used directly for this purpose, since the relationships involved in the fatigue of a ball bearing differ considerably from the fatigue phenomena observed on an ordinary test sample.

Consequently, the test results and practical experience that are available now represent the only data that can be used to establish a calculation procedure. This procedure will have the following purposes:

to establish functions for the service life of bearings under purely radial loads;

to establish rules for the conversion of axial and simultaneously effective axial and radial loads into purely radial loads;

to calculate the effect of different types of loads that are subject to change over time.

Purely radial load. Limited service life is primarily a fatigue phenomenon; however, under exceptionally high loads there will be additional factors such as permanent deformations, direct fractures, and the like. Therefore, to establish a completely correct function for service life requires an extremely great number of tests under different types of loads, r.p.m., bearing dimensions, etc. Unfortunately, it must be admitted that the presently available test results permit only the establishment of a formula of limited validity so that stipulations must be made in the areas that are subject to uncertainty which, however, can be considered justified on the basis of purely practical experience.

If we start out from the assumption that the material has a certain fatigue limit, meaning that it can withstand an unlimited number of cyclic loads on or below a certain, low level of load, the service life curve will be asymptotic. Since, moreover, the material has an elastic and/or fracture limit, the curve must yield a finite load even when there is only a single load value, meaning

that the number of cycles equals zero. If we further assume that the curve has the profile of an exponential function, the general equation for the relationship existing between load and number of load cycles prior to fatigue would read

$$k = C\,(an + e)^X + u \qquad (1)$$

where

k is the specific load,

C is a material constant,

a is the number of load cycles during one revolution at the point with the maximum load exposure within the bearing,

n is the number of revolutions in millions,

e is a material constant that is dependent on the value of the elasticity or fracture limit,

u is the fatigue limit,

X is a certain exponent of negative value.

This general equation for the desired functions should be the simplest expression that will satisfy the imposed requirements and will yield completely applicable results. The constants can be determined with a fair degree of accuracy for high and intermediate loads, but they must become uncertain under very low loads since practical circumstances aggravate the tests within the last-named area. For, under the low loads the data for service life will assume such /340 high values that the required number of tests could not be implemented within the available time, given the available test machinery.

Initially, the specific load was found to be a function of bearing load and bearing dimensions. It is not entirely established that it is also a function of r.p.m., but in practical applications an increased r.p.m. must of necessity result in vibrations and inertia forces in the rotating machinery elements, increasing the actual bearing load. It is a matter of dispute whether this load increase must be taken into consideration immediately when calculating the specific load, and to which degree this must be done, or whether it can be taken into consideration by indirect means through an increase in the dynamic increment added when calculating the effective forces. However, in determining the function for specific load it is best to start out from the equation established by Stribeck,

$$k = \frac{5\,Q}{Z\,d^2} \qquad (2)$$

where

Q is the total radial load on the bearing,

Z is the number of balls in the bearing, and

d is the ball diameter.

However, the constant k requires a certain amount of correction since the effect of the bearing dimensions and probably of the r.p.m. as well, is different from that stated by Eq. (2). The corrected constant, designated k_1 below, will yield good agreement with the tests performed if it is stated as follows:

$$k_1 = k\,(1 + 0{,}0001\,v)\,(1 + 0{,}007\,d^2)\ \text{kg}/(1/8'')^2 \qquad (3)$$

where

k is Stribeck's constant in $\text{kg}/(1/8'')^2$,

v is the r.p.m.,

d is the ball diameter in units of 1/8".

The determination of the material constant e contained in Eq. (1) can be made on the basis of special tests made to determine permanent deformations; these have shown that the specific load for spherical bearings must not be greater than about 40 $\text{kg}/(1/8'')^2$, and not greater than 70 $\text{kg}/(1/8'')^2$ for grooved ball bearings without feed opening and with close contact. The constant C has been determined on the basis of a very great number of test runs under different types of loads. However, certain difficulties are involved in the determination of this constant as a result of the difference in service life demonstrated by the different configurations of the same bearing type under equal test conditions. Therefore, it is necessary to state whether an expression is desired for the minimum, maximum, or for an intermediate service life located between these extremes. It is obvious that it would be very uneconomical in practical applications to dimension all the bearings so that even those will be strong enough which are bound to have a shorter service life. In order to obtain a good, cost-effective result, it is necessary to accept that a certain, small number of bearings will have a shorter service life than the calculated lifetime, and therefore the constants must be selected so that about 90% of all the bearings have a service life longer than stated by the formula. This calculation procedure must be considered entirely satisfactory from both, an engineering and

a business point of view, if we keep in mind that the mean service life is much longer than the calculated service life and that those bearings which have a shorter life actually require only repairs by replacement of the part which is damaged first.

Difficulties that are greater than the ones indicated above are encountered when determining the fatigue limit; it has not been possible as yet to determine whether this fatigue limit is so significant that it will have a decisive effect on the low specific load, or whether it can be disregarded entirely, since it is possible to obtain agreement with experimental data over the entire scope of the tests under both of these assumptions, provided that the other constants, especially the exponent x, are given suitable values.

This exponent x is always located close to 1/3 or 0.3. Its value will approach - 1/3 when the fatigue limit is so high that it cannot be disregarded, and - 0.3 when it is very low. Good agreement with the test results was obtained by stating

$$k_i = \frac{70}{\sqrt[3]{a_n + 5}} + 2 \qquad (4)$$

for double-row spherical SKF bearings, and

$$k_i = \frac{120}{\sqrt[3]{a_n + 5}} + 4 \qquad (5)$$

for single-row grooved SKF bearings without feed opening.

These equations stipulate certain fatigue limits based on several years of experience acquired in practical operation which appears to confirm that the fatigue limit must in fact be taken into consideration. Of course, the material constants contained in the formulae must be referred to a certain type of material so that the validity of the above formulae is confined to the SKF bearing types mentioned above. It is obvious that tests made with bearings of different materials did in fact show that the quality of the material has a very great effect on bearing life.

In order to compute the number of load cycles at the most exposed point in the bearing it is necessary to know the location of this point first. Based on experience acquired here, this point is located in the race of the fixed bearing ring for spherical ball bearings, and in the race of the inner bearing ring for grooved ball bearings. However, this is stated under the prerequisite that the

balls are of extreme quality, which is the case of the SKF bearings. If this is not the case, the balls will represent the weakest element and will render any calculation of the number of cyclic loads in one point impossible. When a ball rolls over a certain point in the race the race material will be exposed to two maximum loads since the greatest exposure of the material is encountered at the edge of the pressure surface. Simple calculations of the motion of the balls in different configurations will yield the following expressions for a:

For spherical ball bearings with rotating inner race

$$a = \frac{2 Z_1}{1 + \frac{r_y}{r_i}} \tag{6}$$

where

Z_1 is the number of balls per row, and

r_y } are the radii of the contact surface of the ball against the { outer race / inner race

r_i }

For spherical and grooved ball bearings with rotating outer race

$$a = \frac{2 Z_1}{1 + \frac{r_i}{r_y}} . \tag{7}$$

For grooved ball bearings with rotating inner race

$$a = \frac{Z_1}{1 + \frac{r_i}{r_y}} \tag{8}$$

Fatigue relationships are subject to some change in the case of <u>axial loads</u> or simultaneously effective <u>axial and radial loads</u> on the bearings. In the case of grooved ball bearings with rotating inner race the number of load cycles will increase with increasing effect of the load acting in the axial direction. However, since all of these loads achieve a maximum only under pure axial load, and since this effect is taken into consideration by the rules formulated for the conversion of axial loads into radial loads, the complicated estimate of the number of maximum loads can be eliminated in this instance.

The life of a ball bearing which is exposed to axial or composite loads is dependent in part on the resulting greater ball pressure, and in part on the fatigue phenomena which have changed in comparison to the relationships existing under purely radial load.

Direct measurements performed on the different bearing configurations[1] are available with respect to the change in ball pressure. It is probably impossible to find an accurate and, at the same time, simple expression for the ball pressure as a function of radial and axial pressure; however, this is hardly required. Adequately precise results can be obtained by using the very simple equation

$$Q = R + yA \tag{9}$$

where

Q is the imagined, purely radial load that will yield the same service /341 life as the simultaneously acting radial and axial forces,

R is the actual radial load,

A is the actual axial load, and

y is a coefficient.

In the case of double-row spherical bearings, y is a function of the so-called compression angle α, that is the angle enclosed by the direction of the pressure by a ball and the radial center plane of the bearing. The function

$$y = \frac{1}{2\sin\alpha} \tag{10}$$

will yield an entirely satisfactory result that agrees with the test data. In the case of bearings of conventional width, y = 3 to 4, and 2 to 3 for wide bearings.

The conversion of the axial forces in grooved ball bearings is much more difficult since we are not dealing with a constant compression angle here; the angle is variable, depending on the load. By calculating the change of the ball pressure and allowing for the elastic deformations as well as the different fatigue configurations which are variable under the different effects of radial and axial pressure, we were able to compile the following series of data for grooved ball bearings without feed opening and with close contact; the y-values were confirmed by the test results:

k in kg/(1/8")^4:	4 to 8	8 to 16	26 to 32	32 to 64
y .....:	1.5	1.2	1.0	0.9

1. cf. Palmgren and Sundberg, Teknisk Tidskrift, Mekanik, Vol. 49 (1919), No. 4, and Palmgren, "Load-carrying capacity of the single-row groovetype ball bearing," Journal of the Society of Automotive Engineers, New York, October 1920.

Variable loads. It can happen in practical operations involving ball bearings that the loads will vary over time, that the r.p.m. is subject to change, or that the load is cyclic within certain intervals. It is very difficult to compute the service life of a bearing under these circumstances, since adequate studies into this problem are still lacking. Gärtner[2] appears to assume that an average load can be used which is obtained through summation of the load products and the number of revolutions during the different periods of time, and dividing the sum by the total number of revolutions. But such an assumption would be erroneous since the service life, as we proved above, is not a first degree function of load.

In order to obtain an approximate value for a calculation, the assumption might be conceivable that a bearing which has a life of n million revolutions under constant load at a certain r.p.m., a portion m/n of its durability will have been consumed. If the bearing is exposed to a certain load for a run of m_1 million revolutions where it has a life of n_1 million revolutions, and to a different load for a run of m_2 million revolutions where it will reach a life of n_2 million revolutions, and so on, we will obtain

$$\frac{m_1}{n_1}+\frac{m_2}{n_2}+\frac{m_3}{n_3}+\cdots\cdots=1 \tag{11}$$

In the event of a cyclic variable load we will obtain a convenient formula by introducing the number of intervals p and designate as m the revolutions in millions that are covered within a single interval. In that case we have

$$p\left(\frac{m_1}{n_1}+\frac{m_2}{n_2}+\frac{m_3}{n_3}+\cdots\cdots\right)=1 \tag{12}$$

where n still designates the total life in millions of revolutions under the load and r.p.m. in question.

Since we now dispose of a much better knowhow on the interrelationships existing between service life and load as well as the other circumstances than we had before, and since we are now able to express these relationships in a simple mathematical form, we have at our disposal a much better means for a dependable and, at the same time, economical determination of bearing dimensions than that provided by the earlier known procedures. That this is indeed true has been proved especially in those areas where antifriction bearings had not been used heretofore so that direct experience was not available.

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2. loc. cit.

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